

# Considerations regarding a thermopower installation having a diesel engine with hydrogen combustion, and steam turbine

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## ABSTRACT

In this paper the authors make a comparative analysis of thermopower installations operated with hydrogen and oxygen, installations appropriate to cover the daily peak loads in a power system. Comparing the type of installation having rotating motors and a gas turbine or steam turbine with an installation format of diesel engine and steam turbine, we argue the latter alternative as preferable.

Here, we discuss in more detail the installation with diesel engine in which the oxygen and hydrogen obtained from water electrolysis are recombined. In order to utilize completely the steam energy resulting from this recombination after it is released from the diesel engine, this energy is used in a steam turbine with condensation. In this way the entire installation has an effective efficiency about 50% larger as compared with the former alternative cited above.

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## 1. Introduction

Described in [1] was an installation designed to deliver electrical energy (EE) in a period of the year when the demand curve in a power system (PS) registers maximum value. In that installation during the “peak load,” periods an additional EE based on the hydrogen (H) combustion previously obtained by electrolysis, is produced. The EE needed for the electrolysis process is produced in a nuclear power plant (NPP), in that period when EE is in less demand for PS. In that case, we had discussed more powerful Energy Plants (330 MW and 700 MW).

It is known that in a PS, the energy demand fluctuates on a large scale during a year period, but there exist more reduced fluctuations during every day. These yearly and daily fluctuations are illustrated by the very known “load curves”. In the daily load curve case, during a 24 h period, maximum values are registered during the 8:00–10:00 am and 5:00–7:00 pm (intervals), respectively. Fig. 1 represents, as an example, the load curve of one day of the year 2004 as registered in the Romanian National Power System (RNPS) case [2].

On the ordinate axis at the left of the figure, we represent the power in MW of RNPS needed to satisfy the demand of EE (curve no. 1). This power was provided by the NPP of Cernavoda, marked between the arrows with no. 2, by the Thermo Power Plants operating with Coal (TPPC) – space 3–, by the hydro power plants (HPP) –

space 4–. The remainder up to curve 1 is provided by the Thermo Power Plants operating with Hydrocarbons (TPPH) – space 5–.

For technical reasons, NPPs, TPPC and the majority of HPPs must operate as much as possible with a constant load and as close as possible to the nominal power. The power variation in PSs, and especially in RNPS, is provided by TPPHs and some HPPs. This ensured with EE of a PS during the periods when the demand is increasing, in some cases very much and fast, constitutes a difficult problem, especially when, in that PS, a big role of EE production belongs to the NPPs and HPPs, which operate without interruption.

Presently in Romania, the specific weight NPP for Cernavoda (2x700 MW) in RNPS represents about 9% of the installed weight.

On the other side, the legal norms to reduce atmosphere pollution will determine an important increase of unconventional energy sources, especially of wind and solar energy. EE obtained from these sources, by its fluctuating nature and direct use in majority of the cases, cannot be included in the variable portion of the load curve. Thus, for example, it is known that the solar energy has its maximum value at noon during sunny days. In Fig. 1, the curve drawn with a fragmented line represents the solar energy variation during that corresponding day as a percentage expressed with reference to the ordinate axis at the right of figure [3]. Also, the wind energy exhibits large fluctuations, a fact that follows from the analysis dedicated to this problem in [4]. In order to accumulate EE during those periods when its production is greater than the demand, so that the accumulated energy can be delivered during the peak load periods, many different technical systems have been conceived. For PSs of large capacity these systems also

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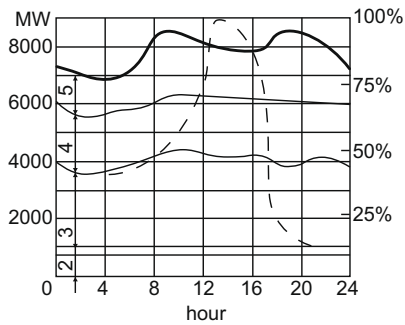


Fig. 1. The power load curve of Romanian Power System for a day from 2004.

generate powers with large values. For example, in Germany, the installation from Huntorf [5] is equipped with a gas turbine (GT) of 290 MW which operates an electric generator (EG) and an air compressor (AC). The generator and compressor have the capacity for rapid coupling and decoupling among themselves. In these periods when there is a surplus of EE in (the) PS, the combination EG–AC is decoupled by the GT, and the EG has a role of an electrical motor which drive(s) the AC. The compressed air at a pressure of 47.5 bars is stored in a salt reserve cave. During the peak load periods, the EG is decoupled from (the) AC and coupled with the GT. The compressed air needed to operate the GT is obtained from the “reservoir” of the salt reserve cave and the GT–EG combination delivers EE in (the) PS.

In Japan, another country with a well developed PS, the first hydro-power unit of 470 MW “pumped storage” HPP (PSHPP Kannagawa) [6] was placed in operation at the end of 2005. During the periods of minimum consumption in the PS, the electrical generator (EG) functions as an electrical motor, and the hydraulic-turbine (HT) functions as a pump. During the peak load period, the hydraulic processes are performed inversely; HT is fed with water from the superior accumulation lake and the water is evacuated in the inferior lake; EG delivers EE in PS. In PSHPP Kannagawa, there will be installed another six units of 470 MW by 2016.

In Romania studies were made for PSHPP Tarnita-Lapustesti with a total power between 500 MW and 1000 MW. Performance will be optimal after introducing in RNPS of the third unit of 700 MW to the NPP Cernavoda probably after 2015.

In the Romanian situation, in order to secure the delivery of the additional EE during the daily peak load periods, there is, in the near future, a need for power installations with powers of less than 50 MW. Given ecological demands, it is imperative that these installations operate with a minimum contribution to atmospheric pollution.

As we have shown in [1], EE additionally produced from the demand in some periods can be used to produce hydrogen (H) which, in its turn, can be used to produce EE in a thermopower installation (as needed). In this way, we can assign value to the qualities of H as “energy agent” [7,24].

In order for these thermopower installations (TPIs) to respond to this purpose, they must be simple and very fast to start operation with relative small power, at least for the RNPS case.

In the following chapter, we will comparatively analyze the possible varieties of such TPIs.

## 2. The thermopower installations operated with hydrogen

During the H combustion in a TPI (in the boiler, combustion chamber, etc.), when the air has the carrier role, none of the toxic gases or greenhouse gases such as  $\text{SO}_2$ , CO or  $\text{CO}_2$  are produced; however, nitrogen oxides ( $\text{NO}_x$ ) are produced and these are toxic, also having a greenhouse effect.

In what follows, we (will assume the production of H by water electrolysis, a process in which oxygen (O) is also produced, and we will refer to the TPIs using the steam as working fluid. (That is), H and O resulting from electrolysis will recombine (stoichiometric) by combustion, and steam will result which, in its turn, will produce mechanical energy in a thermal motor and the corresponding EE in the electric generator operated by this motor. Consequently, the TPI is totally non-polluting.

If we refer to the TPIs in any thermal motor with steam turbine (ST), gas turbine (GT) or diesel engine (DE), then mechanical work is produced by working fluid expansion from a high value pressure to (a lower) value. In GT and DE, which are using the classical conventional fuels such as hydrocarbons, the fluid gas expansion is pushed up to the atmospheric pressure or higher since the gas evacuation is done in the atmosphere directly or else by thermal recovery. For STs a solution was found to increase the working fluid steam expansion with pressures much lower than atmospheric pressure. This is realized by steam evacuation at its turbine exit in a condenser in which the steam is condensed with the help of a cooling fluid (water or air) brought from an exterior place. In this way, a closed circuit of the working fluid is created (water–air). In principal, the closed circuit is made by steam boiler, ST, condenser and condensate pumps inserted with feeding pumps of the boiler. The operation of such a thermopower installation from a thermodynamic point of view is inscribed in a well known Rankine cycle.

Referring to Fig. 2, in the case of installations with STs, the Rankine cycle consists in the following thermodynamics processes represented on  $T$  (absolute temperature scale)– $s$  (entropy).

- (1) The transportation towards the boiler and the increase of the water temperature resulting from the steam condensation taken over from the condenser by the condensate pumps and further by the feed water pumps – segment ab.
- (2) The water heating – segment bc – its transformation in steam – segment cd – and steam superheating – segment de – can be considered as an example of an overheated steam installation. These processes are produced in the steam boiler.
- (3) The adiabatic–isentropic steam expansion (the theoretical case represented in Fig. 2) in ST – segment ef.
- (4) The steam condensation in the condenser – segment fa.

In this case, the recombination of H and O results in steam as the working fluid as we have shown for the other thermal motors (GT and DE, respectively). In order to obtain the highest mechanical work, we can apply the pressure reduction system during the evacuation by introducing a condenser in the installation. In this way, the thermodynamic processes in the TPIs equipped with such types of machines can be inscribed in a thermodynamic cycle similar to Rankine cycle, with the following particularities:

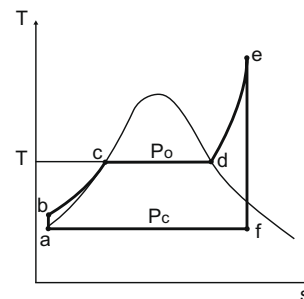


Fig. 2. The Rankine cycle represented in  $T$ – $s$  coordinates.

- The process represented by the segment ab has the purpose to feed the electrolyzer (replacing the boiler) with water.
- The electrolytic process (is indicated by) the point b of the diagram in Fig. 2.
- The combustion process in the combustion chamber of GT or in the cylinders of DE replaces the water heating, vaporization and the steam overheating. In this way, the line bcde of Fig. 2 represents, virtually only, the real processes that have place in the case of the installations with GT or DE.

The TPIs with GT or DE using classical fuels (hydrocarbons) evacuate noxious gases with and prevent toxic greenhouse effects. The TPIs functioning with the recombination of H with O, would evacuate water vapor into the atmosphere and could be considered totally non-polluting. In the future, there is also a possibility that the water vapor emissions into the atmosphere could be considered to be noxious. Many scientists and scientific institutions regard water vapor as elements of the greenhouse gas category [8]. Regarding this aspect, the installations with condenser (systems) operating in conformance to the cycles described above have more advantages since water-steam is included in the thermodynamic processes that operate in a closed circuit.

In order to make comparisons among the TPIs equipped with thermal motors mentioned above (ST, GT or DE), from the point of view of their utilization in satisfying the daily load curve during the peak loads, we will accept that the pressure of the steam evacuation in all cases to be 0.04 bar. This is the present value of the pressure obtained in the condensers of the steam turbine using cooling water at 15 °C.

It is known that the thermodynamic theoretical efficiency of a Rankine cycle (Fig. 2) is given by the formula:

$$\eta_{tt} = (i_e - i_f) / (i_e - i_a) \quad (1)$$

where  $i_e$ ,  $i_f$  and  $i_a$  are the values for the steam enthalpies in the respective points (e, f, a) in Fig. 2. The points e and f denote the beginning and respectively the end of the steam adiabatic-isentropic expansion. Thus the pressure  $p_0$  and temperature  $T_0$  represent the steam parameters at the inlet in the thermal motor, and  $p_c$  represents the pressure in the condenser.

In Fig. 3 we represent the variation curves of the efficiency  $\eta_{tt}$  for different values of  $p_0$  and  $T_0$  in the domain where it is commonly used for the thermal motors ( $p_{0\max} = 200$  bar,  $T_{0\max} = 2000$  K). For  $p_c$  in all cases, we accepted  $p_c = 0.04$  bar. Consequently, for  $i_a$  we accept the corresponding values for  $p_c = 0.04$  bar and  $i_a = 118.27$  kJ/kg.

We mention that to avoid complications in our comparative analysis, the relation (1) and the traced curves in Fig. 3 based upon them refer to the most simple TPIs. These are not provided with systems to improve the efficiency such as regenerated preheating,

intermediate overheating or adoption of supercritical parameters ( $p_0$  and  $T_0$ ), as in STs case, or to other heat recovery systems such as GTs and DEs, respectively.

In the majority of the cases, the parameters of these TPIs join the domains mentioned below and can be identified in Fig. 3. From this figure also results the maximum values for  $\eta_{tt}$  in the usual cases for the corresponding installations. Thus:

- For TPIs with ST – ( $p_0 = 15$ –186 bar); ( $T_0 = 520$ –808 K) [5]...  $\eta_{tt\max} = 0.45$ .
- For TPIs with GT – ( $p_0 = 10$ –20 bar); ( $T_0 = 1200$ –1773 K) [9]...  $\eta_{tt\max} = 0.51$ .
- For TPIs with DE – ( $p_0 = 40$ –170 bar); ( $T_0 = 1300$ –2500 K) [10,11]...  $\eta_{tt\max} = 0.62$ .

The maximum unit power for the motors cases analyses, realized up to the present, has the following values: 1100 MW, for conventional TPPs [12] and respectively, 1500 MW, for NPPs [13], in the STs case, about 280 MW [14] in GTs case, and about 70 MW in the DEs case [15].

Another important element in our analyses is the starting time (along with the time needed to reach) nominal power which characterize each of the compared installations. This characteristic has values of (1–4) h, in the STs case, (10–30) min, in the GTs case and (15–60) s, in the DEs case.

Based upon the above data, we conclude that the most efficient alternatives to furnish EE in RNP during the daily peak load periods are TPIs with DEs, given stoichiometric combustion of H having O as carrier. Such a type of TPI would not pollute and would have, in comparison with the other analyzed TPIs, the following advantageous particularities: maximum efficiency, minimum starting time, and nominal power reaching. The unit power, though not very large, would correspond also to our mentioned purpose.

In what follows, we will analyze in more detail such a TPI with DE operated with H.

### 3. The thermopower installations with diesel engines functioning with H, and with steam turbine

The thermopower installation which we intend to analyze is represented schematically in Fig. 4. Specified are the most important components and devices from which it is formed.

In Fig. 4, the corresponding circuits of diverse fluids are represented as such: H – double fragmented line; O – broken line accompanied by a continuous line; steam – double continuous line; water – continuous simple line. With a simple discontinued line, we represent the electrical connection for continuous current – dc between the electrical energy source (EES) 3 and the electro-motive 2.

Since the installation comprises also the electrolyser where H and O are produced, we name it power installation with diesel engine and hydrogen combustion (abbr. PIDEH). This installation is characterized by two operating stages depending on the load of the power system (PS), namely:

4.1. During these periods of PS reduced load, with the help of the Intermediate Pressure Pump (IPP17), a quantity of water is pumped in the electrolyser EL2. H and O produced in EL2 are stored in the hydrogen tank 4 and in the oxygen tank 6, respectively. In this operation stage, the valves 20, 21 and 23 are open and the valves 19, 22 and 24 are closed. The water is pumped with the help of IPP17 in such a way that H and O are stored in the tanks 4 and 6 at a pressure of  $p_{IP\max} = 50$  bar. A quantity of water from tank 1 is used to “moderate,” that is, to regulate the fluid temperatures (H, O, water, steam) in the thermodynamic processes.

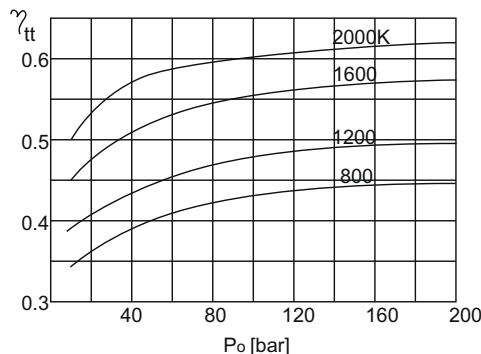
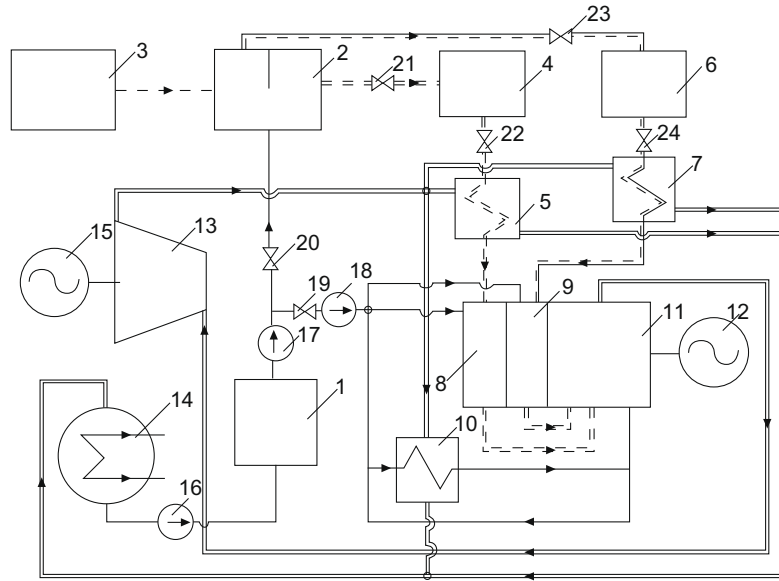


Fig. 3. The Thermodynamic theoretical efficiencies for Rankine cycles.

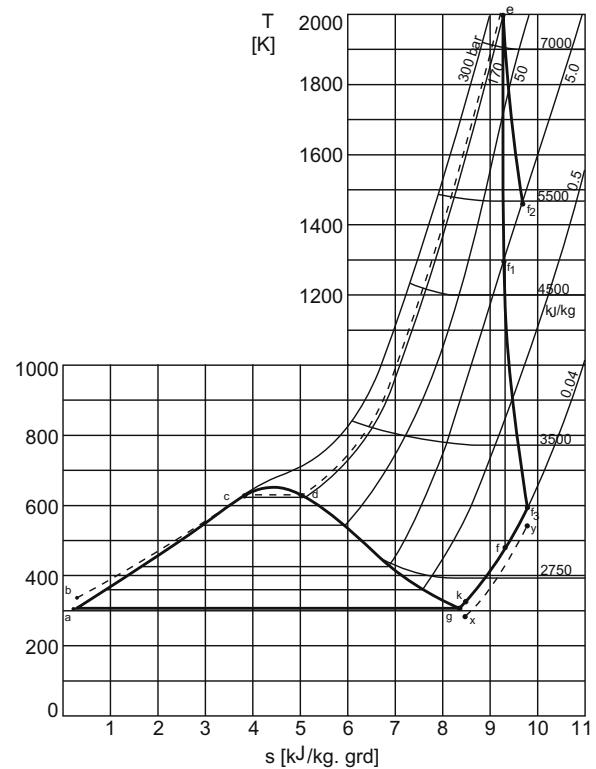


**Fig. 4.** The PIDEH scheme. 1. Water tank; 2. Electrolyser; 3. Electrical energy source; 4. Hydrogen (H) tank; 5. H preheater; 6. Oxygen (O) tank; 7. O preheater; 8. Reciprocating Compressor; 9. Reciprocating Compressor; 10. Tempering water preheater; 11. Diesel Engine (DE); 12. Electrical Generator for DE; 13. Steam Turbine (ST); 14. Steam Condensator; 15. Electrical Generator for ST; 16. Condensate Pump; 17. Intermediate Pressure water Pump (IPP); 18. High Pressure water Pump (HPP); 19. Water valve; 20. Water valve; 21. Hydrogen valve; 22. Hydrogen (pressure reduction) valve; 23. Oxygen valve; 24. Oxygen (pressure reduction) valve.

4.2. During the peak load periods, PIDEH has an active role in EE production. In this stage, the valves 19, 22 and 24 are open and the valves 20, 21 and 23 are closed. We mention that in the scheme of the Fig. 4 representation, only the valves which are strictly needed to explain the PIDEH operation during these two distinct stages are indicated. The valves 22 and 24 are in the same time pressure regulators having the role to reduce the pressure from the tanks 4 and 6 up to the value of  $p_{ac} = 25$  bar, at which H and respectively O are introduced into the Reciprocating Compressors 8 and 9. In this way, the H and O quantities disposed will correspond to the decreasing values of the pressure in the tanks 4 and 6 from 50 bars to 25 bar.

In order to increase the installation efficiency in the tanks 4 and 6 and in the compressors 8 and 9, H and O pass through the preheaters, PHs 5 and 7, respectively. In these PHs, the two gases are preheated with the help of the steam evacuated from ST13. At the entrance in the PHs, this steam is still overheated, as we can see in the Fig. 5. After PHs 5 and 7, the steam arrives at the condenser 14. From this condenser, the water is transported with the help of condensate pump 16 to the tank 1. In the compressor RC9, the oxygen is compressed further up to the pressure  $p_{dO} = 170$  bar, and in RC8 the hydrogen is compressed up to the pressure  $p_{dH} = 190$  bar. Both compressors 8 and 9 are directly driven by the diesel engine (DE) 11. This engine also drives the electric generator (EG) 12. Before the DE with piston (for simplification we consider that both RCs and DE are composed each of one cylinder and correspondingly, one piston) gets to the Top Dead Center (TDC), with an adequate admission advance, O from RC9 is admitted into the DE at a pressure of 170 bars. As we have shown, O has a carrier role. In that moment when the piston passes through the TDC, H is “injected” into the cylinder with a pressure of 190 bar. The process of introducing H into the cylinder is conducted in a way that it produces a constant pressure combustion on a portion of about  $40^\circ$  of one  $180^\circ$  rotation which is performed by the engine crankshaft corresponding to the piston stroke between TDC and Bottom Dead Center (BDC) [11].

By combustion (H and O combination), steam is produced which is “tempered” by injecting water such that the steam tem-



**Fig. 5.** The thermodynamic cycle of PIDEH represented in  $T$ - $s$  diagram.

perature will reach a temperature of 2000 K and the pressure of 170 bar. This is expanded in the DE cylinder up to the pressure of evacuation (5 bars). Further, the steam enters into ST 13, where it is expanded up to the pressure  $p_c = 0.04$  bar. In this way, we can see that the thermodynamic cycle of DE 11 becomes an “Atkinson cycle” [16], which is characterized in using the working fluid evacuated from the DE for the further producing of EE by its expansion in GT (in our case ST).



The steam expansion processes in DE 11 and ST 13 are represented in the Fig. 5 diagram:  $T$  (absolute temperature)– $s$  (entropy).

The theoretical expansion (adiabatic–isentropic) in DE 11 is represented by the  $ef_1$  line and the real expansion (polytropic) is represented by the  $f_1f_3$  line. More detailed explanations regarding these processes will be given in the next chapter.

For a complete recovery of the heat contained in the evacuated steam from ST 13, a portion of this heat is used as a heating fluid in PH 10 in order to preheat a portion of the water used for tempering in DE 11, that is, to achieve the maximum desired temperature in D11 of 2000 K. The rest of the water needed in this operation bypassed PH 10 and it blends with the preheated water before its introduction in DE 11.

We mention that, in other situations, if it needed, water injection is also used for tempering the fluid gases in GTs [17].

In these three PHs (5), (7), and (10), the heat contained in the evacuated steam from ST 13 is recovered along the line  $f_3k$  before it passes to the cooling stage in the condenser and condensation (line  $kga$ ).

Again, in order to increase the PIDEH efficiency we also use “the isothermal compression” of H and O with the help of RCs 8 and 9. In doing that, we used the results given in [11]. In this paper, an installation with a DE (“isoengine”) coupled with a compressor having a piston is described, where an isothermal compression, at a constant temperature, of the air used for the natural gas fuel combustion is realized.

The entire quantity of the thermal energy, which in our case, needs to be “extracted” from the two compressed gases (H and O) in order to obtain their compression at a constant temperature, remains in the tempering water introduced in RCs 8 and 9. This water is introduced in DE 11 at the same time with H and O, and thus the heat accumulated in the compression process is recovered at the level of the entire PIDEH.

From the same article [11], we adopted some values of the PIDEH diverse characteristics such as, the fuel and carrier pressures, the pressures at the evacuation from DE and respectively the admission in ST, the maximum temperature in DE, etc. These values were already mentioned in a previous description of installation operation. Our chosen values for other measures, in order to further present the performed calculations, are the following:

- The ambient medium temperature (cooling air and water),  $t_0 = 15^\circ\text{C}$  ( $T_0 = 288\text{ K}$ ). The same value temperature was accepted also for the water in the tank 1, for H in the tank 4 and for O in the tank 6.
- The pressure of the water discharge from the high pressure pump HPP18,  $p_{hp} = 250\text{ bar}$ . In this way the water injection to obtain the isotherm compression in RC8, RC9 and the tempering in DE11 is obtained. In this case, IPP17 has the role of a booster pump for HPP18.

The thermodynamic processes mentioned above, in their progression are a cycle which is partially similar to a Rankine cycle. In Fig. 5, it is represented by the lines denoted abcdefga for the theoretical cycle with the adiabatic–isentropic steam expansion in DE11 and ST13. In the real case (polytropic expansion), the corresponding cycle is abcdef<sub>2</sub>f<sub>1</sub>f<sub>3</sub>ga. As we have shown previously in Fig. 2 and also in the classical situation, the portion bcde in the  $T$ – $s$  diagram represents the processes of the water heating and boiling and further the steam overheating in the boiler. In our case, we consider that on the portion ab where the water pressure increase is produced with the help of IPP17 and the bb segment which represents the electrolysis process in EL2 as well as H and O compression up to the pressures 190 bar and 170 bar, respectively. The steam is produced in DE11 only and that is represented by bcde line. In regard to the diagram  $T$ – $s$  for water–steam, these

processes can be considered virtual only and consequently we represent them with a fragmented line. The processes of the steam expansion (theoretic – the line ef and effectively – the segmented line ef<sub>2</sub>f<sub>1</sub>f<sub>3</sub>), the steam cooling (the line f<sub>3</sub>g) and its condensation (the line ga) are inscribed in a classic Rankine cycle.

#### 4. The effective efficiency of PIDEH

Let us proceed to calculate the effective efficiency of the real cycle for PIDEH.

For this we need to begin to determine the thermodynamic characteristics of the steam expansion in DE and ST, that is, the steam temperature values and enthalpies at the  $f_1$ ,  $f_2$ ,  $f_3$  and  $f$  points of the  $T$ – $s$  diagram represented in Fig. 5.

The theoretical steam expansion (adiabatic–isentropic) is represented by the line  $ef_1$ , and the real expansion is represented by the line  $ef_2$  in DE. In ST, the theoretical expansion is represented by the line  $f_1f$ , and the real expansion is represented by the line  $f_1f_3$ . For the line  $f_2f_1$ , we will give explanations latter.

In Fig. 5, we see that the temperature in the point  $f_1$  is  $T_{f1} = 1300\text{ K}$ . If we apply the steam conditions for  $p_{de} = 5\text{ bar}$  and  $T_{f1}$  mentioned above, we obtain the steam enthalpy  $i_{f1} = 4799\text{ kJ/kg}$ . In the same way calculated, the initial steam enthalpy is  $i_e = 7420\text{ kJ/kg}$  is obtained. In order to calculate the steam temperature and enthalpy at the point  $f_2$ , we apply the relation for the internal efficiency of the steam expansion in the DE, where, by its analogy with the efficiency expression of the thermal rotating motors (ST, GT), the formula is:

$$\eta_{iE} = (i_e - i_{f2}) / (i_e - i_{f1}) \quad (2)$$

Generally, with internal combustion motors with piston (Reciprocating Engines) and thus also in DEs, an internal efficiency similar to the above mentioned is not used, but instead, some other efficiencies as such as the incomplete combustion efficiency, the indicated efficiency ( $\eta_{in}$ ), the volumetric efficiency or more precisely the fullness degree ( $\eta_v$ ) [16]. We denoted the incomplete combustion efficiency by the symbol,  $\xi$ , as in the case of the installations with GTs [17]. In [10], the “plenitude degree”  $\eta_p$  is defined and this indicates the combustion perfection as well as the thermal losses. In the DE case,  $\eta_p = 0.8$ – $0.85$ . This efficiency value is best for a comparison with the efficiencies of the thermal rotating motors, which indicate the same types of thermodynamic losses. Thus in the GT case we can say that the incomplete combustion in the combustion chamber and thus the effective efficiency is given by the relation:

$$\eta_{eGT} = \xi_T \cdot \eta_{iT} \cdot \eta_{mT}, \quad (3)$$

where  $\xi_T$  is the efficiency or the degree of the incomplete combustion,  $\eta_{iT}$  is the internal efficiency and  $\eta_{mT}$  is the mechanic efficiency of the turbine. By comparison, for the DE case we can write:

$$\eta_{eE} = \eta_p \cdot \eta_{mE}, \quad (4)$$

where  $\eta_{mE}$  is the mechanic efficiency of DE.

Using the definition of  $\eta_p$  given above, we can write:

$$\eta_p = \xi_E \cdot \eta_{iE}, \quad (5)$$

where  $\eta_{iE}$ , referring to our case, is given by the relation (2). The degree of the incomplete combustion in the specialty literature is indicated by  $\xi_T = 0.96$ – $0.98$  [17] and  $\xi_E = 0.95$  [18]. For our future calculations, we choose  $\xi_E = 1.0$  since in DE, a stoichiometric recombination of H and O is produced. The mechanical efficiency of the turbines have values in the domain of  $\eta_{mT} = 0.92$ – $0.99$  [5]. We choose  $\eta_{mT} = 0.95$ . In the DE case, the mechanical efficiency has reduced values since it control the losses determined by the admission and the evacuation of the motor fluid in the cylinders [16]. It

has limit values  $\eta_{mT} = 0.8\text{--}0.9$  [18]. We choose  $\eta_{mE} = 0.85$ . If we accept  $\eta_p = 0.80$  with the value for  $\xi_E$  accepted above, we obtain  $\eta_{IE} = 0.8$  from relation (5).

With the help of relation (2), we obtain  $i_{T2} = 5323.2$  kJ/kg. It results in  $T_{T2} = 1469$  K.

The internal mechanical work of DE is:  $L_{IE} = i_e - i_{T2} = 2096.8$  kJ/kg.

For STs and GTs, the thermodynamic losses conducting to  $\eta_{IT} < 1$  are transformed in heat which remains in the interior of the respective machine. These losses are manifested by the enthalpy and respectively by the temperature increase of the working fluid at the end of its expansion compared with the situation when this expansion could be done without internal losses such as adiabatic–isentropic losses. Unlike STs and GTs for DE, the cooling process of the mechanical parts in contact with heated gases intervenes (cylinder, piston, cylinders heads, valves). The heat transferred to these parts is evacuated to the exterior by the cooling water and oil. If, for simplification, we assume that the entire heat resulting from DE internal losses is evacuated to the exterior by its cooling systems, the result is that the steam parameters at the evacuation from the DE are the ones corresponding to the point  $f_1$  of the diagram  $T$ – $s$ . The theoretic process (adiabatic–isentropic) of the steam expansion in ST is represented by the line  $f_1f$ . The point  $f$  is situated on the isobar  $p_c = 0.04$  bar. The temperature on the point  $f$  is  $T_f = 473$  K and the enthalpy is  $i_f = 2863.3$  kJ/kg. For the internal efficiency of ST a similar relation with relation (2) is:

$$\eta_{IT} = (i_{f1} - i_{f3}) / (i_{f1} - i_f). \quad (6)$$

The efficiency  $\eta_{IT}$  has values in the domain  $\eta_{IT} = 0.85\text{--}0.89$  [5]. We choose  $\eta_{IT} = 0.87$ . With the help of the relation (6), we obtain  $i_{f3} = 3114.94$  kJ/kg. We get  $T_{f3} = 594$  K.

Referring in DE, to a quantity of 1 kg of the working fluid (steam) for the entire thermodynamic cycle of PIDEH, the effective efficiency is given by the relation

$$\eta_e = (L_E + L_T - L_{CH} - L_{CO}) / \zeta \cdot Q_{IH} \quad (7)$$

where,  $L_E$  is the effective mechanical work produced by DE,  $L_T$  is the effective mechanical work produced by ST,  $L_{CH}$  is the effective mechanical work consumed by the hydrogen compressor (RCH) and  $L_{CO}$  is the effective mechanical work consumed by the oxygen compressor (RCO). At the fraction numerator,  $\zeta$  (kgH/kgw) represents the quantity of H consumed as fuel in DE corresponding to 1 kg of water ( $w$ ) in the form of steam, which is the motor fluid in DE and ST.  $Q_{IH}$  is the inferior caloric power of H.

In order to calculate the hydrogen quantity  $\zeta$  and the corresponding O quantity, we apply the known chemical equation, which characterizes the water decomposition process (in our case by electrolysis in EL) and the recombination of H and O (in our case by combustion in DE):



Quantitatively, if we change from units in moles to units in kg, electrolysis results in 2 kg of H and 16 kg of O from 18 kg of water, or from 9 kg of water is produced 1 kg of H and 8 kg of O, and inversely in the combustion case.

From the PIDEH operation description, we remember that the water quantity which becomes the working fluid (under the steam form) in DE and ST is composed from:

- $w_1$  – the quantity of water decomposed in H and O by electrolysis and recomposed in DE. Expressed as a function of  $\zeta$  we have:

$$w_1 = 9 \cdot \zeta \quad (9)$$

- $w_2$  – the quantity of water used for the H compression tempering, thus this cooling process should be isothermally produced.

- $w_3$  – the quantity of water used for cooling the O compression tempering, thus this cooling should be isothermally produced.
- $w_4$  – the preheated water quantity (in PH10) used for tempering in DE.
- $w_5$  – the non preheated water quantity, used together with  $w_4$  for tempering in DE.

Evidently, when we refer to 1 kg of working fluid, we have:

$$w_1 + w_2 + w_3 + w_4 + w_5 = 1 \quad (10)$$

As we can see in Fig. 4, H and O are preheated in PHs 5 and 7 before being compressed in RC 8 and RC9. The heat is taken over by the steam evacuated from ST at a pressure  $p_c = 0.04$  bar and a temperature  $T_{f3} = 594$  K, as previously determined.

We accept that H and O are preheated at a temperature  $T_0 = 288$  K existent in the tanks 4 and 6, up to the temperature,  $T_{ach} = T_{aco} = 554$  K. We consider that a temperature difference between the steam at the entrance in PH 5 and PH 7 and H and O at the exit from the pre heaters is  $DT = 40$  grd is needed to transfer the heat in PHs. That is,  $T_{ach} = T_{aco} = T_{f3} - 40$ .

The theoretic mechanical work consumed by RC 8 in the isothermal compression case, considering H as a perfect gas, is given by the relation:

$$L_{CHt} = \zeta \cdot R_H \cdot T_{ach} \cdot \ln(p_{dch}/p_{ach}) \quad (11)$$

Let  $\zeta$  (kg) be the quantity of H (from one kg of water), as we have shown, and  $R_H$  is the gas constant for H which is  $R_H = 4.1217$  kJ/kg grd. Let the temperature  $T_{ach} = T_{dch}$  (the isothermal compression) and the pressures  $p_{dch}$  (at the evacuation from RC 8) and  $p_{ach}$  (at the aspiration in RC 8) have the above mentioned values. Then  $T_{ach} = T_{dch} = 554$  K,  $p_{dch} = 190$  bar and  $p_{ach} = 25$  bar. Introducing these values into the relation (11), we obtain  $L_{CHt} = 4631.12 \cdot \zeta$  (kJ/kg).

Similarly, the theoretical mechanical work consumed by RC 9, considering O as a perfect gas, is given by relation:

$$L_{COt} = 8\zeta \cdot R_O \cdot T_{aco} \cdot \ln(p_{dco}/p_{aco}), \quad (12)$$

where,  $8\zeta$  (kg) is the quantity of O (corresponding to  $\zeta$  quantity of H),  $R_O = 0.2598$  kJ/kg grd,  $T_{aco} = 554$  K,  $p_{dco} = 170$  bar and  $p_{aco} = 25$  bar. We obtain  $L_{COt} = 2207.21 \cdot \zeta$  (kJ/kg).

In order to obtain an isothermal compression in RC8 and RC9, the water injected there has to take over from the H and O which are compressing. The thermal energy is exact equivalent of the mechanical work,  $L_{CHt}$  and,  $L_{COt}$ , respectively. Let us denote by  $Q_{w2}$  and  $Q_{w3}$  the respective heat quantities (by  $w_2$  and  $w_3$  indicating the referring water quantities), then we get:

$$Q_{w2} = w_2 \cdot c_w \cdot (T_{ach} - T_0) = L_{CHt} \quad (13)$$

where  $c_w$  is the specific water temperature, considered constant,  $c_w = 4.1868$  kJ/kg grd,  $T_{ach} = T_{dch} = 554$  K and  $T_0 = 288$  K.

Introducing these values in the relation (13) and the value of  $L_{CHt}$  calculated above we obtain  $w_2 = 4.15836 \cdot \zeta$ . Similarly, we will have:

$$Q_{w3} = w_3 \cdot c_w \cdot (T_{aco} - T_0) = L_{COt} \quad (14)$$

Having  $T_{aco} = T_{dco} = 554$  K, results  $w_3 = 1.9819 \cdot \zeta$ .

We assume that the heat exchange in the pre heaters HPH5 and OPH7 is performed at a constant temperature change of  $\Delta T = 40$  grd. This means that the steam temperature at the end of the preheating of H and O is  $T_k = 288 + 40 = 328$  K. The point  $k$  of the diagram  $T$ – $s$  (Fig. 5) expresses the steam parameters at the exit from HPH5 and OPH7, and the point  $f_3$  represents the steam parameters at their entrance. By the fragmented line  $xy$  we represented the heating processes of H and O. The point  $x$  represents the parameters at the entrance of these gases in the preheaters and  $y$  represents the parameters at the exit of the gases from the pre-

heaters. In  $x$  we thus have  $T_0 = 288$  K and in  $y$  we will have  $T_y = T_{acH} = T_{acO} = 554$  K. In order to calculate the taking over temperature in HPH5 and OPH7 of the evacuated steam from ST, we calculated its enthalpy value at the point  $k$  ( $p_c = 0.04$  bar;  $T_k = 328$  K). Then we have  $i_k = 2602.5$  kJ/kg.

From the relations (13) and (14), we get  $Q_{w2} = 4631.12 \cdot \zeta$  and  $Q_{w3} = 2207.21 \cdot \zeta$ . The quantities of the heat taken over by the steam along the line  $f_3k$ , namely  $Q'_{w2}$  and  $Q'_{w3}$ , are equal to the ones above ( $Q_{w2}$  and  $Q_{w3}$ ) divided by the heat exchange efficiency  $\eta_{he}$ . For the heat exchanger in the “countercurrent,” these efficiencies generally are values in the domain  $\eta_{he} = 0.85$ – $0.95$ , [19]. We choose  $\eta_{he} = 0.90$ . Thus, we will have  $Q'_{w2} = Q_{w2}/\eta_{he} = 5145.56 \cdot \zeta$  and  $Q'_{w3} = Q_{w3}/\eta_{he} = 2452.46\zeta$ .

Knowing that from ST, 1 kg of steam is evacuated, the remaining heat in the evacuated steam from ST, after the retaking over of the quantities  $Q'_{w2}$  and  $Q'_{w3}$ , is:

$$Q'_{w4} = i_{f3} - i_k - Q'_{w2} - Q'_{w3} \quad (15)$$

Introducing the intervening values in the right side of the equality (15), we obtain

$Q'_{w4} = 512.44 - 7598.02 \cdot \zeta$ . The heat taken over by the water quantity  $w_4$  is:

$$Q_{w4} = \eta_{he} \cdot Q'_{w4} = 461.196 - 6838.218 \cdot \zeta \quad (16)$$

On the other side

$$Q_{w4} = w_4 \cdot c_w \cdot (T_y - T_0) = 1113.69 \cdot w_4 \quad (17)$$

From (16) and (17) we have  $w_4 = 0.414 - 6.14 \cdot \zeta$ .

The material balance is expressed in the relation (10). Introducing in this relation the quantities  $w_1$  to  $w_5$  expressed as a function of  $\zeta$ , we have:

$$9 \cdot \zeta - 0.584 + w_5 = 0 \quad (18)$$

The Eq. (18) contains two unknowns,  $\zeta$  and  $w_5$ . In order to determine them, we need another equation, which contains these two unknowns. This second equation is from the energy balance result with reference to DE (for 1 kg of working fluid). This has the form:

$$I_H + i_0 + i_{w2} + i_{w3} + i_{w4} + i_{w5} + q_{ch} = i_e \quad (19)$$

where,

- $i_H$  is the enthalpy (in kJ, referring finally to 1 kg of working fluid) of H introduced in DE. (For all enthalpies from below is valid the same mention about the dimension – kJ –).

$$i_H = \zeta \cdot c_{pH} \cdot T_{dch} \quad (20)$$

For  $T_{dch} = 554$  K, the specific heat of H is  $c_{pH} = 14.581$  kJ/kg grd and thus  $i_H = 8077.87 \cdot \zeta$ .

- $i_0$  is the enthalpy of O introduced in DE.

$$i_0 = 8 \cdot \zeta \cdot c_{pO} \cdot T_{dcO} \quad (21)$$

For  $T_{dcO} = T_{dch} = 554$  K, the specific heat of O is  $c_{pO} = 1.024$  kJ/kg grd and thus  $i_0 = 4538.37 \cdot \zeta$ .

- $i_{w2}$  is the water enthalpy for tempering of H during its compression, introduced under the form of heat in DE at the same time with H.

$$i_{w2} = w_2 \cdot c_w \cdot T_{dch} = 9645.26 \cdot \zeta \quad (22)$$

- $i_{w3}$  is the water enthalpy for tempering of O during its compression. It is introduced in the form of heat in the DE at the same time as O.

$$i_{w3} = w_3 \cdot c_w \cdot T_{dcO} = 4596.99 \cdot \zeta \quad (23)$$

- $i_{w4}$  is the preheated tempering water enthalpy introduced in DE.

$$i_{w4} = w_4 \cdot c_w \cdot T_y \quad (24)$$

Replacing  $w_4$  from (17) in (24), we obtain  $i_{w4} = 960.267 - 14241.65 \cdot \zeta$ .

- $i_{w5}$  is the non preheated tempering water enthalpy introduced in DE.

$$i_{w5} = w_5 \cdot c_w \cdot T_0 = 1205.8 \cdot w_5 \quad (25)$$

- $q_{ch}$  is the heat introduced in DE by combustion of a  $\zeta$  quantity of H.

$$q_{ch} = \zeta \cdot Q_{iH} \quad (26)$$

where  $Q_{iH}$  is the inferior caloric power of H. Having  $Q_{iH} = 119617$  kJ/kg, we obtain  $q_{ch} = 119617 \cdot \zeta$ .

Introducing in (19) the above calculated enthalpies, and assuming that  $i_e = 7420$  kJ/kg, and performing the addition operations in order to compare it with the relation (18), we obtain the second equation containing  $\zeta$  and  $w_5$ , namely:

$$109.66 \cdot \zeta - 5.357 + w_5 = 0 \quad (27)$$

From (18) and (27), we have  $\zeta = 0.0474$  kg and  $w_5 = 0.1594$  kg.

In order to calculate the effective efficiency of PIDEH, we return to the relation (7), specifying the following:

- $L_E$  (the effective mechanical work produced in DE) is given by the relation:

$$L_E = \eta_{mE} \cdot L_{iE} = \eta_{mE} \cdot (i_e - i_{f2}) \quad (28)$$

where  $\eta_{mE}$  is the mechanical efficiency of DE, accepted previously as having the value  $\eta_{mE} = 0.85$ , and  $L_{iE}$  is the interior mechanical work of DE. With the values of  $i_e$  and  $i_{f2}$  as determined previously, we have  $L_E = 1782.28$  kJ/kg.

- $L_T$  (the effective mechanical work produced by ST) is given by the relation

$$L_T = \eta_{mT} \cdot L_{iT} = \eta_{mT} \cdot (i_{f1} - i_{f3}) \quad (29)$$

where  $\eta_{mT}$  is the mechanical efficiency of ST, as previously accepted as having the value  $\eta_{mT} = 0.95$ .

$L_{iT}$  is the internal mechanical work of ST. With the values of  $i_{f1}$  and  $i_{f3}$  previously determined we have  $L_T = 1599.88$  kJ/kg.

- $L_{CH}$  (the effective mechanical work used to drive the compressor of H) is given by the relation

$$L_{CH} = L_{cHt}/\eta_{mCH} \quad (30)$$

where  $L_{cHt}$  is the theoretic mechanical work used to drive this compressor and  $\eta_{mCH}$  is the mechanical efficiency of the compressor.

- $L_{CO}$  (the effective mechanical work used to drive the compressor of O) is given by the relation

$$L_{CO} = L_{cOt}/\eta_{mCO} \quad (31)$$

where  $L_{cOt}$  is the theoretic mechanical work used to drive this compressor, and  $\eta_{mCO}$  is the mechanical efficiency of the compressor.

For the values of the mechanical efficiency of the compressor with piston, it is indicated, that the domain  $\eta_{mCO} = 0.85$ – $0.90$  [20]. We accept  $\eta_{mCH} = \eta_{mCO} = 0.90$ . From previous calculations, we have  $L_{cHt} = 4631.12 \cdot \zeta$  and  $L_{cOt} = 2207.21 \cdot \zeta$ . Introducing the value found for  $\zeta$ , we have  $L_{cHt} = 219.51$  kJ/kg and  $L_{cOt} = 104.62$  kJ/kg. With these value we obtain  $L_{CH} = 243.9$  kJ/kg and  $L_{CO} = 116.25$  kJ/kg.

kg. Introducing into relation (7) all these values given above and knowing that  $Q_{\text{H}} = 119617 \text{ kJ/kg}$ , we obtain  $\eta_e = 0.533$ . We mentioned that in all calculations mentioned above, in order to determine the value of  $\eta_e$ , we did not take into consideration the pressure losses in the PIDEH circuits. In any case these have minimum values compared with other thermodynamics losses, which were properly considered. We appreciate that these are equivalent with about 2% points in the effective efficiency value and thus this will not have a value smaller than  $\eta_e = 0.51$ . The theoretic effective efficiency value of a simple Rankine cycle (without heat recovering), for the initial steam parameters similar with those we previously accepted ( $p_0 = 170 \text{ bar}$ ,  $T_0 = 2000 \text{ K}$ ), is  $\eta_{\text{tt}} = 0.62$  (Fig. 3).

It is important to notice that for the regular diesel motors, in the professional literature [18,21] there are indications that the effective efficiencies values should be in the domain  $\eta_e = (0.3\text{--}0.4)$ . As a superior limit for this efficiency which we intend to reach in the “isoengines” case, it is indicated that the value of  $\eta_e$  could be  $\eta_e = 0.6$  [11].

Finally, we also must mention that in this paper, we did not tackle the concrete use of H in motors with internal combustion where some complex phenomenon intervenes such as the admission processes, ignition by compression, combustion, etc. These problems are entirely special and are discussed as such in the professional literature [21–23,25,26].

## 5. Conclusions

From our discussion in the previous chapters, we can make some important conclusions:

5.1. Generally, we know that in a power system (PS), during the time periods with reduced loads, a surplus of electrical energy (EE) is produced and in the time periods with peak loads there is a need of an additional (EE) in comparison with what was produced in PS. There exist technological systems, which insure the needed “transfer” of EE between the corresponding periods.

In this paper we consider the system, relative to these daily electrical load variations in PS, during the periods in which the surplus of EE, hydrogen (H) and oxygen (O) is produced by electrolysis. These gases are temporally stored and are recombined in the thermopower installations with thermal motors producing the supplementary EE in the peak load periods.

This system has the advantage that while producing EE no noxious or greenhouse gases are produced since the H combustion is done stoichiometrically with O having a role as carrier.

Another advantage comes from the fact that H and O are produced by electrolysis using as much as possible non-polluting systems in the production of EE, as is the case for nuclear power plants (NPPs), the hydro power plants (HPPs) or systems using the primary unconventional energy (wind, solar, etc.).

5.2. An installation was devised so that during the reduced load periods in PS, H and O are produced by electrolysis. These gases are stored to be used during the peak load periods in a PS. During these periods H and O are recombined by a stoichiometrical combustion process and steam is produced. The steam, after it partially expands in DE, is further expanded in a steam turbine with condensation (ST). In this way, the pressure difference increases during the steam expansion. The expansion is produced up to a pressure less than the atmospheric pressure (0.04 bar) and consequently obtains a superior mechanical work relative to that obtained by the expansion to the atmospheric pressure only.

For an additional increase of the thermodynamic cycle efficiency, we apply the solution of the isothermal compression of H and O in the compressors with pistons driven by DE (which at the same time drive an electrical generator-EG). Also, we recover the heat contained in the steam evacuated from ST (which drives in its turn another EG) before this steam is allowed to condense. Thus, we obtain an installation which operates in a closed circuit (water-steam-water), with no impact (not even steam emission) on the environment.

5.3. The effective efficiency of the installation described in 5.2 above is about 50% larger than in the energy systems with GT, ST, or conventional diesel motor installations.

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